JTX 9

Sundstrand Power Systems

Sundstrand Power Systems / TURBOMECA
APS-3200 APU
COORDINATION MEMO

MEMO No. ST-1493
DATE: 1/29/93
REPLY BY: 2/5/93

TO: G. Hardy FROM: T. MAEDCHE / P.J SUTTIE

SUBJECT: LOAD COMPRESSOR CONTROL

REFERENCE:
The following referenced information is [ ] (or is not [ ]) considered "PROPRIETARY" by the originator

[X] REQUEST
[ ] INFORMATION
[ ] REPLY TO:

SPS is looking in detail at the Load Compressor Control and some questions have arisen. Please review this data and provide assistance as to how the B factor can be modified to prevent this situation arising

Engine: Q25
Location: SPS Tail Section
ECS Configuration: FCV 1 = Open
FCV 2 = Open
Facility Valves Open

IGV Position: 40% Constant

The problem sequence can be described in the following three steps:

1. Bleed on commanded -

IGV's Go to 40%
Delta P/P = .14
Bc = 2.65, B=2.723 (LCD Temp = 173.23 DegC)

State-

B>Bc and Delta P/P < Set Point

The ECB correctly commands the BCV to its mid-position thus dumping bleed flow down thee surge duct.

2. LCD temperature increases until -

B=2.605 (LCD Temp. = 179.928 degrees C)
Now B<Bc

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APPROVED BY: [signature] DATE: 2/1/93

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HSB 035001

DEPOSITION EXHIBIT 232

PENGAD-Bayonne, N.J.

BCV is commanded full open to load

Note: State two is only momentary - to the point at which the BCV moves

3. BCV is at its full open to load position

o Having obtained the data corresponding to this scenario, SPS requests the "B Factor" calculation be revised to prevent delays or loss of flow to the customer.

DATA-

State 1

Delta P = 3.019 Psid
Pstatic = 20.4 Psid
B=2.723
LCD Temp = 173.23 Deg C

State 3

Delta P = 15.143 Psid
Pstatic = 32.472 Psid
B=2.605
LCD Temp = 179.928 Deg C

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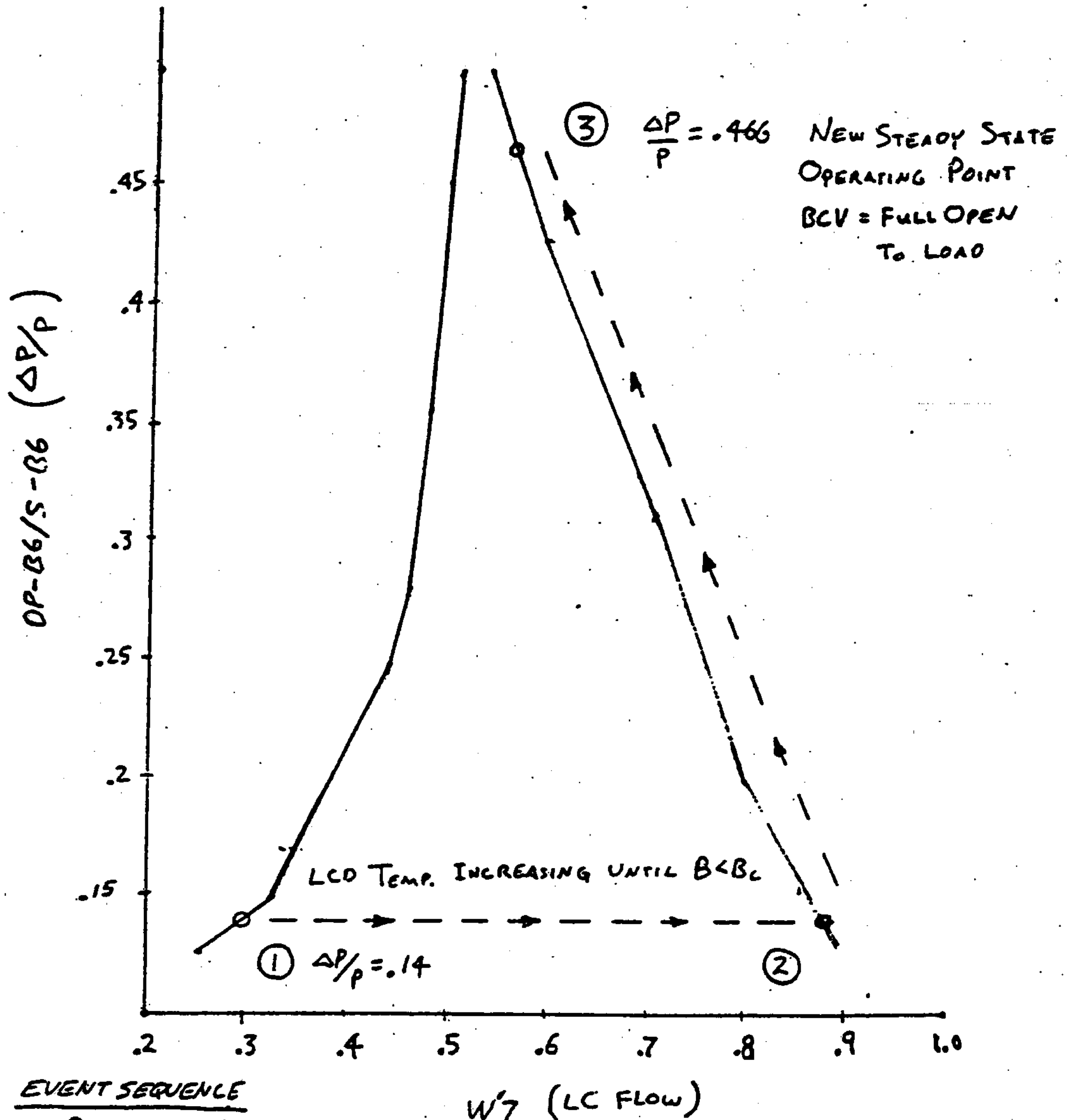
MEMO No. ST-1493



— ECB CURVE
o ΔP/p DATA PTS FROM ECB

EVENT SEQUENCE

① B > B_c

①→② LCD Temp. Increases

② B < B_c (DUE TO LCD TEMP. CHANGE) ∴ BCV ALLOWED TO OPEN

②→③ BCV OPENS TO LOAD

③ NEW STEADY STATE OPERATING POINT

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file

Sundstrand Power Systems / TURBOMECA
APS-3200 APU
COORDINATION MEMO

Sundstrand Power Systems
MEMO No. ST- 1619
DATE: 10 Mar 93
REPLY BY:

TO: G. Hardy FROM: PETE SUTTIE

SUBJECT: UNDERSTANDING THE B FACTOR

REFERENCE:

The following referenced information is [X] (or is not [ ]) considered "PROPRIETARY" by the originator

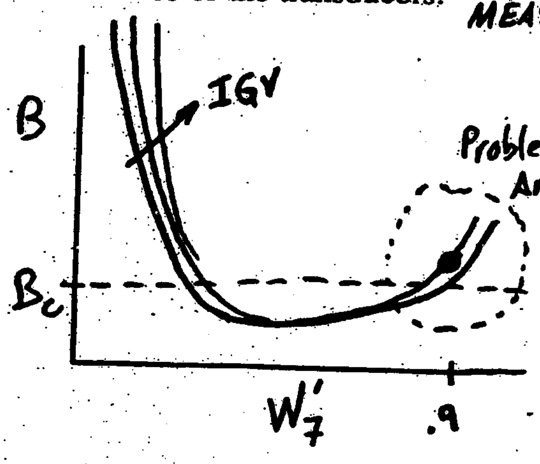
[ ] REQUEST
[X] INFORMATION
[ ] REPLY TO:

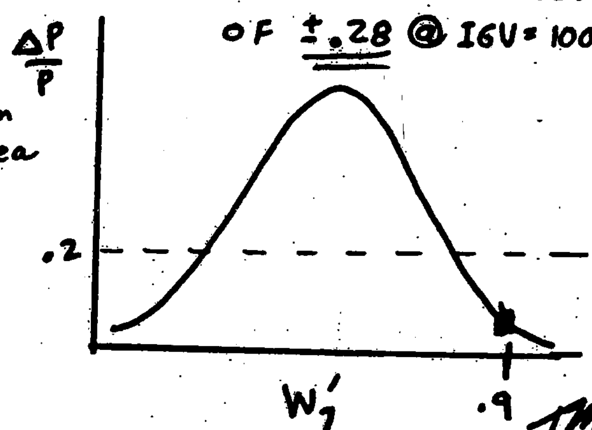
Having reviewed coord memo TS-332-0580 dated 26 Nov 91, I can offer an explanation and solution as to why the "B Factor" is preventing the control system to open fully to load during high flow conditions (i.e., W Bleed > 100 PPM and ΔP/P< set point).

If you refer to my rough sketch, I believe the problem occurs at high flows when the calculated value of B is very close to B critical. Consequently, if B should happen to be above B critical, and ΔP/P is below the set point, then the ECB will correctly command the BCV to its mid-point. The problem can "self correct" only if B calculated goes below BC for an instant, and the BVC moves slightly. This changes the pressure ratio and causes ΔP/P to "jump" above the set point.

The problems associated with the bleed factor appear to have been solved by raising the values of B critical. However, I am questioning if the values selected take into account the tolerances of the transducers.

MEASUREMENT OF B HAS A TOLLERANCE OF ±.28 @ IGV = 100%





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Sundstrand Power Systems / TURBOMECA
APS-3200 APU
COORDINATION MEMO

MEMO No. ST-1782
DATE: 27 May 93
REPLY BY: 1 Jul 93

TO: G. Hardy FROM: Ed Edelman

SUBJECT: B-Factor Control Logic Problems

REFERENCE:
The following referenced information is ☒ (or is not ☐) considered "PROPRIETARY" by the originator

☒ REQUEST
☐ INFORMATION
☒ REPLY TO: TS 332-0580
TS 332-0623
TS 332-1200

The attached memo details why the B factor, as it has presently been defined is not acceptable for surge control.

The memo discusses BCV Control problems experienced both at Turbomeca and SPS.

The problem with the B Factor is that it does not allow for inaccuracies in the sensors. The sensor accuracy values are given in Appendix B.

SPS requests that Turbomeca urgently define a better method of determining which side of the ΔP/P curve the load compressor is operating on.

SPS SUGGESTIONS

1) Providing a tolerance on the be value which allows for the sensor tolerances.

2) Redefining Bc without LCDT.

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APPROVED BY: [signature] DATE: 6/2/93

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EXHIBIT
62
Edelman

HSB 035443

Power Systems SUNDSTRAND AEROSPACE

memo

DATE: May 27, 1993

TO: Distribution

FROM: Ed Edelman

REF: ECE:072:052693

SUBJECT: B-Factor Control Logic Problems

cc: S. Lampe
M. McArthur
K. Mehr-Ayin
P. Suttie

Problem:

The following surge control problems are a result of the B-Factor, as defined by Turbomeca (reference Coord. Memo TS 332-1200).

1. Incorrect prediction of low load compressor airflow by the critical B-Factor (Bc) results in full bypass airflow when not required. This condition is often misdiagnosed as a bleed control valve (BCV) sticking problem, and translates into poor customer flow performance.

2. Sensor inaccuracies were not factored into the selection of the critical B-Factor (Bc).

3. Load compressor discharge temperature (LCDT) rise after selection of bleed results in a temporary full bypass flow condition (5-10 seconds).

Recommendations:

1. Turbomeca must identify the maximum Bc bias required for sensor variation and associated worst case condition for control (i.e., minimum $\Delta P/P$) throughout the load compressor operating envelope (see *"Sensor Variation not considered in determination of Bc"* section below), or:

   Provide all operating conditions where B = Bc to SPS for sensor tolerance analysis.

2. Turbomeca must redefine Bc without LCDT included, or provide sufficient Bc margin to offset LCDT dynamic affects.

   This is required to prevent the shift in the B-Factor from a predicted low flow to high flow condition immediately after a bleed command (see *"Load compressor discharge temperature (LCDT) dynamics"* section below).

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B-Factor Unreliable for surge control

The critical B-Factor (Bc), which is designed to differentiate between low and high load compressor flow, is unreliable, and results in uncommanded full bypass flow at certain operational conditions.

The B-factor (B) is used by the load compressor surge control, where

$$B = \frac{P7 - \Delta P}{PT2} \left(\frac{TT2}{T7 - TT2} \right)$$

where P7 is load compressor discharge pressure
T7 is load compressor discharge temperature
TT2 is inlet temperature
PT2 is inlet pressure
ΔP is the pressure difference between the load compressor scroll and diffuser

If B≤ Bc, high air flow is assumed (and no surge control protection is required). If B > Bc, than low air flow is assumed, and active closed loop control is invoked, as shown in Figure 1. Bc is a function of IGV angle, as shown in Table 1.

IGV	BC
0	1.35
22.9	2.40
43.7	3.20
64.6	3.35
85.4	3.50
95.8	3.20
100.0	3.10

Table 1: Bc, as a function of IGV position. The table was updated per TS332-1200.

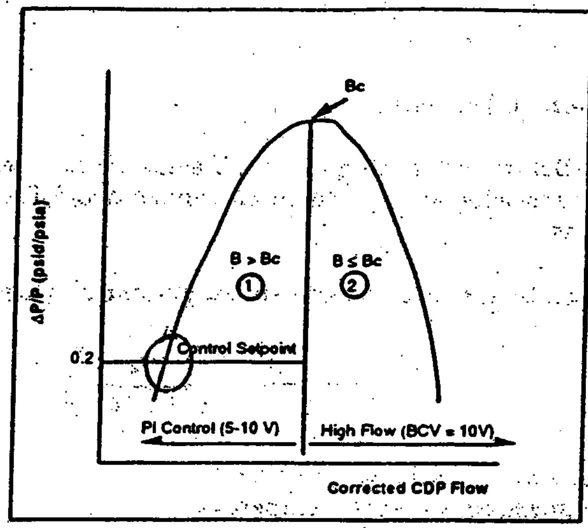

Figure 1: B-Factor operation. If B> Bc (region 1), surge control is active. If B ≤ Bc, high flow is assumed, and the BCV is commanded to 10V, or full customer flow.

The control problem arises when the B-Factor incorrectly calculates a low condition (Region 1, Figure 1) when a high flow condition is present (Region 2, Figure 1), and $\Delta P/P < 0.20$. Since the slope of the $\Delta P/P$ vs. W7c is reversed, positive feedback occurs, and the BCV commands 5V, or maximum surge bypass flow.

B-Factor modified by Turbomeca, control anomaly persists

Bc was modified by Turbomeca per Coordination memo TS332-1200 (see Appendix A). However, testing showed that the Bc selected still results in false diagnosis of a high flow condition.

Control logic modified to provide B-Factor robustness

The following control logic was implemented to provide B-factor robustness, and prevent false diagnosis of a high flow condition.

If $\Delta P/P > 0.35$, the control assumes a high flow condition. Thus, the control logic is active in region 1 of Figure 2. Hysteresis bands were added to the B-factor calculation and $\Delta P/P$

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HSB 035446 4/10

calculation to prevent BCV cycling experienced with B-factor shifting during steady-state operation, which resulted in 5 to 10 V oscillations.

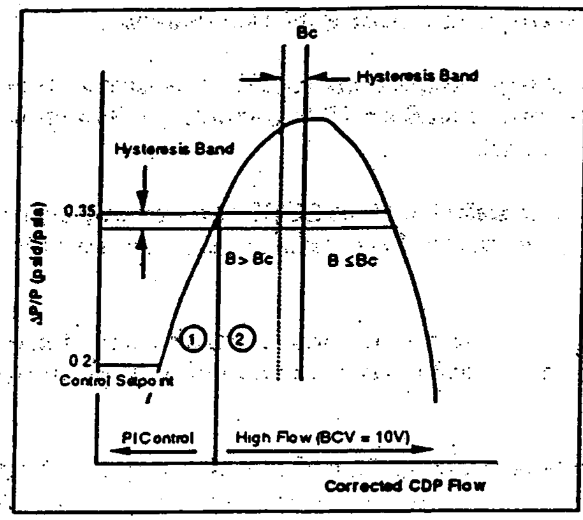

Figure 2: Surge control is active when B > Bc and ΔP/P <0.35, or Region ①. The ΔP/P criteria was added as a temporary software solution to prevent bleed valve cycling due to incorrect Bc.

Control logic modification results in reduced dynamic surge control protection

Implementation of the ΔP/P = 0.35 criteria reduces the capability of the BCV closed-loop surge control to anticipate an impending surge condition.

With the ΔP/P setpoint (0.20) in close proximity to the high flow setpoint (0.35), the PI control has limited time to react, thereby reducing dynamic surge control performance.

Sensor Variation not considered in determination of Bc

Sensor variation was not considered in determining Bc.

A preliminary study (Appendix B) based on one operating condition identified a ~~0.25~~ 0.23 bias to Bc is required.

Turbomeca must identify the maximum Bc bias required for sensor variation and associated worst case condition for control (i.e., minimum ΔP/P) throughout the load compressor operating envelope.

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Load compressor discharge temperature (LCDT) dynamics

Load compressor discharge temperature (LCDT) rise after a bleed command results in a temporary full bypass flow condition (5-10 seconds), as shown in Figure 3 (normal operation) and 4 (abnormal operation).

An increase of approximately 100 °F in LCDT resulted in a shift in the B-Factor from a predicted low flow to high flow condition. This results in an unacceptable (and unpredictable delay) in customer bleed flow.

Introducing a lead-lag term to the Bc control setpoint would not improve reliability, because of the LCDT time constant variability as a function of back pressure due to ECS (or MES) flow restriction.

I recommend Turbomeca redefine Bc without LCDT included, and determine the resultant variability in ΔP/P.

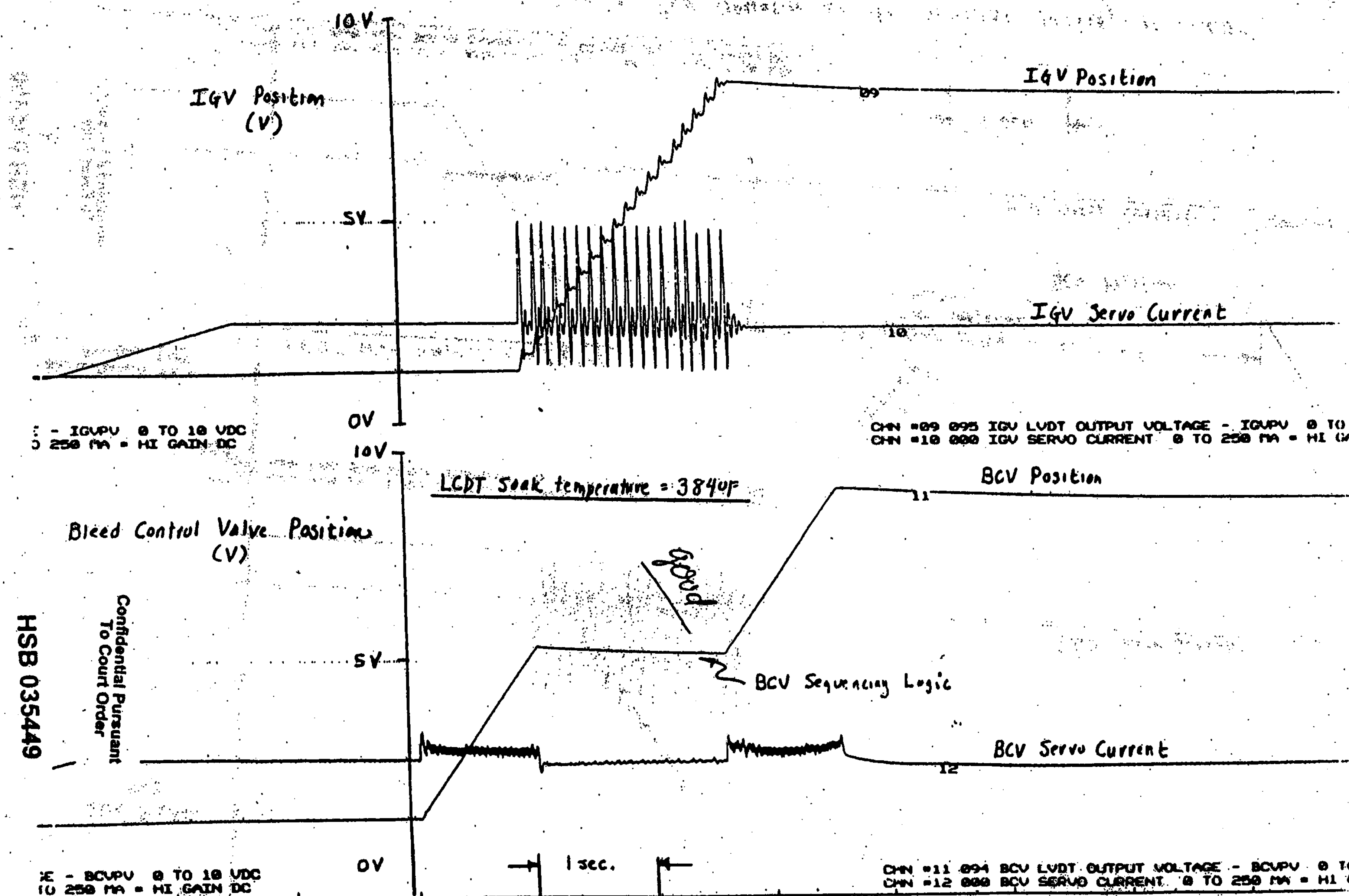

Figure 3: Normal Operation of the Bleed Control Valve and IGV Sequencing Logic after a Bleed Command.

7/D

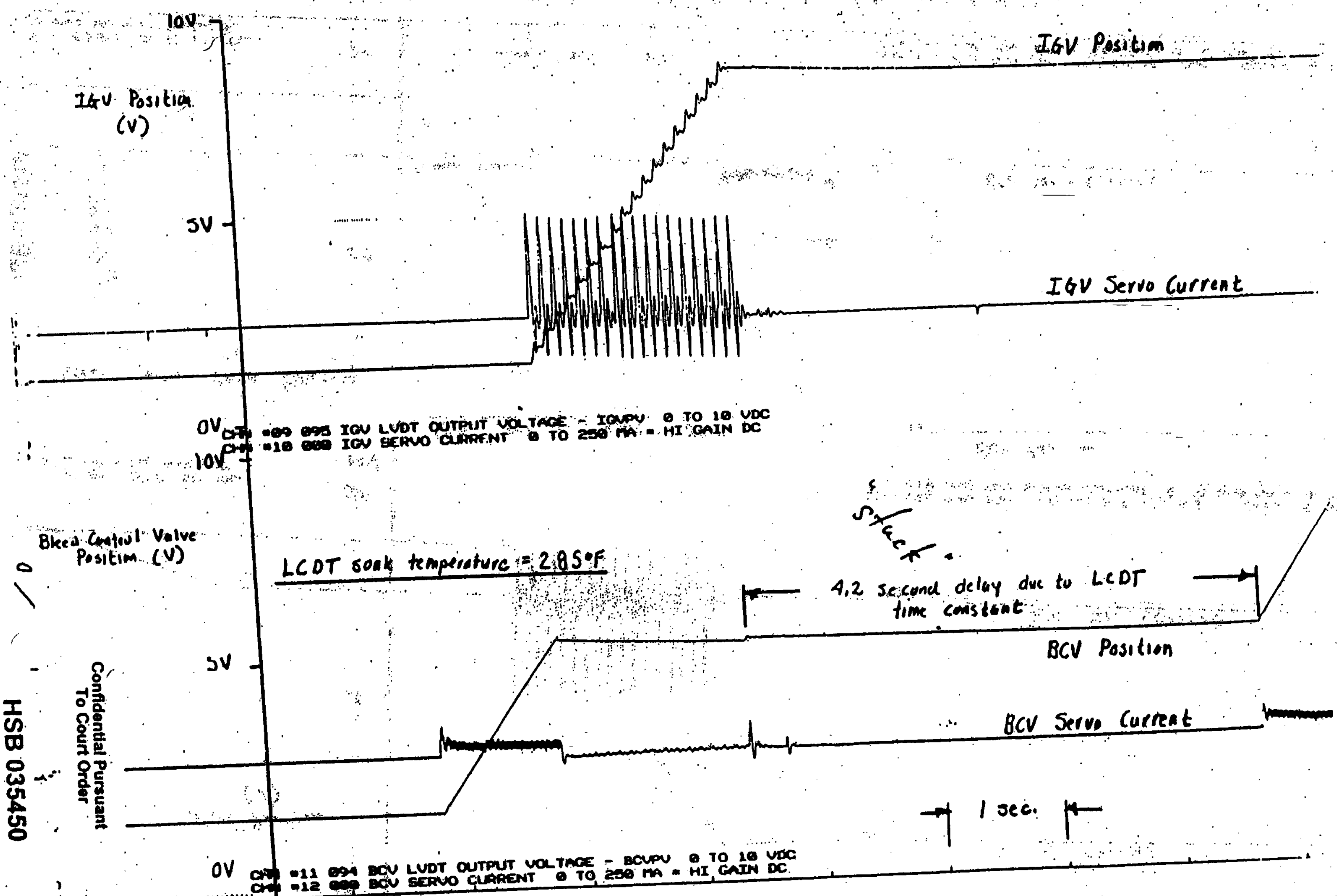

Figure 4: Abnormal BCV Operation due to transient heating of LCDT.

APS3200 May 26, 1993 APPENDIX B B-Factor Sensor Tolerance EDELMAN, ED 1/2

Problem: Determine B_c tolerance stack-up due to sensor inaccuracies. PROBL

Given: GIVEN

i) Sensor Tolerance:

$P_7 = \pm 3.5\%$
$\Delta P = \pm 3.5\%$
$P_{inlet} = \pm 3.5\%$
$T_{inlet} = \pm 5°F$
$T_7 = \pm 6°F$
$IGV = \pm 1.5\%$

ii) Typical $B \approx B_c$ operating condition

$P_7 = 42.08$ psia
$T_7 = 384$ °F
$P_{inlet} = 14.04$ psia
$T_{inlet} = 71.2$ °F
$\Delta P = 18.64$ psia
$IGV = 90.6\%$

iii) $B = \dfrac{P_7 - \Delta P}{P_{inlet}} \left(\dfrac{T_{inlet}}{T_7 - T_{inlet}} \right)$

Find: Least-squares ΔB tolerance; F.ND

$$\Delta B = \sqrt{\Delta B_{P_7}^2 + \Delta B_{P_{inlet}}^2 + \Delta B_{T_7}^2 + \Delta B_{T_{inlet}}^2 + \Delta B_{IGV}^2 + \Delta B_{\Delta P}^2}$$

$$B = \frac{42.08 - 18.64}{14.04} \left(\frac{71.2 + 459.688}{384 - 71.2} \right) = 2.84$$ ANAL

$$B + \Delta B_{P_7} = \frac{(42.08 + 1.47) - 18.64}{14.04} \left(\frac{71.2 + 459.688}{384 - 71.2} \right) = 3.01$$

$\Delta B_{P_7} = 0.17$ ← ΔB_{P_7}

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HSB 035451

Analysis, cont.

$$B + \Delta B_{T7} = \frac{42.08 - 18.64}{14.04}\left(\frac{71.2 + 459.69}{384 + 6 - 71.2}\right) = 2.78$$

$\Delta B_{T7} = 0.06$ ← ΔB_{T7}.

$$B + \Delta B_{P\,inlet} = \frac{42.08 - 18.64}{14.04 + 0.49}\left(\frac{71.2 + 459.69}{384 - 71.2}\right) = 2.74$$

$\Delta B_{P\,inlet} = 0.102$ ← ΔB_{P_2}

$$B + \Delta B_{T\,inlet} = \frac{42.08 - 18.64}{14.04}\left(\frac{71.2 + 5 + 459.69}{384 - 71.2 + 5}\right) = 2.82$$

$\Delta B_{T\,inlet} = 0.02$ ← ΔB_{T_2}

$$B + \Delta B_{\Delta p} = \frac{42.08 - 18.64 + 0.65}{14.04}(1.697) = 2.75$$

$\Delta B_{\Delta p} = 0.09$ ← $\Delta B_{\Delta P}$

$B_c + \Delta B_{c\,IGV} = 90.6\% + 1.5\% = 92.1\%$

$$\frac{\Delta B}{\Delta IGV} = 0.0288$$

$\Delta B = 0.0288(1.5) = 0.043$

$\Delta B_{IGV} = 0.043$ ← ΔB_{IGV}

$$\Delta B_{total} = \sqrt{0.17^2 + 0.06^2 + 0.102^2 + 0.02^2 + 0.09^2 + 0.043^2} = 0.23$$

ΔB_{total}

$\Delta B_{total} = 0.23$ ←

10/10

HSB 035452

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Sundstrand Power Systems

Sundstrand Power Systems / TURBOMECA

APS-3200 APU

COORDINATION MEMO

MEMO No. ST-2377

DATE: 3 Nov 94

REPLY BY: 4 Nov 94

[x] REQUEST

[ ] INFORMATION

[ ] REPLY TO:

TO: L. Brayle, P. Biscay FROM: E. Edelman

SUBJECT: Load Compressor Control Design - LC Pressure Ratio as Flow Predictor

REFERENCE:

The following referenced information is [ ] (or is not [ ]) considered "PROPRIETARY" by the originator

As we discussed 20 October at Aerospatiale, it would be advantageous to replace the B-factor with load compressor pressure ratio P7/P2 as a function of corrected speed and IGV position for prediction of choked flow vs. low flow conditions for the surge control. This predictor would only be required for $\Delta P/P \leq 0.35$ psid/psia.

This would eliminate the requirement for using the Load Compressor Discharge Temperature (LCDT) sensor. As you are aware, the slow time constant associated with this sensor results in miscalculation of the B-factor, resulting in full-bypass of the bleed control valve, contributing to the low bleed pressure during MES.

The alternative is a T7 hardware change to a fast acting thermocouple, which is undesirable.

This is a critical problem that must be resolved for V3.0. Lilian, as discussed in Today's Telecon, Deutsche Airbus is concerned that our design review has been delayed one week and we will not meet the V3.0 deadline as a result.

A rapid response is necessary to meet the V3.0 deadline. It is the only remaining design issue to date. A simple response specifying the required measured parameters would be sufficient. The P7/P2 value(s) could be supplied at a later time.

If possible, could you provide this simple response tomorrow? I understand this is an important design issue, and takes proper analysis to complete.

Best Wishes,

Ed Edelman

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EXHIBIT
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JTX 28

Aug. 1, 1961 G. M. LEWIS ET AL 2,994,471

AIR SUPPLY UNIT

Filed March 10, 1958

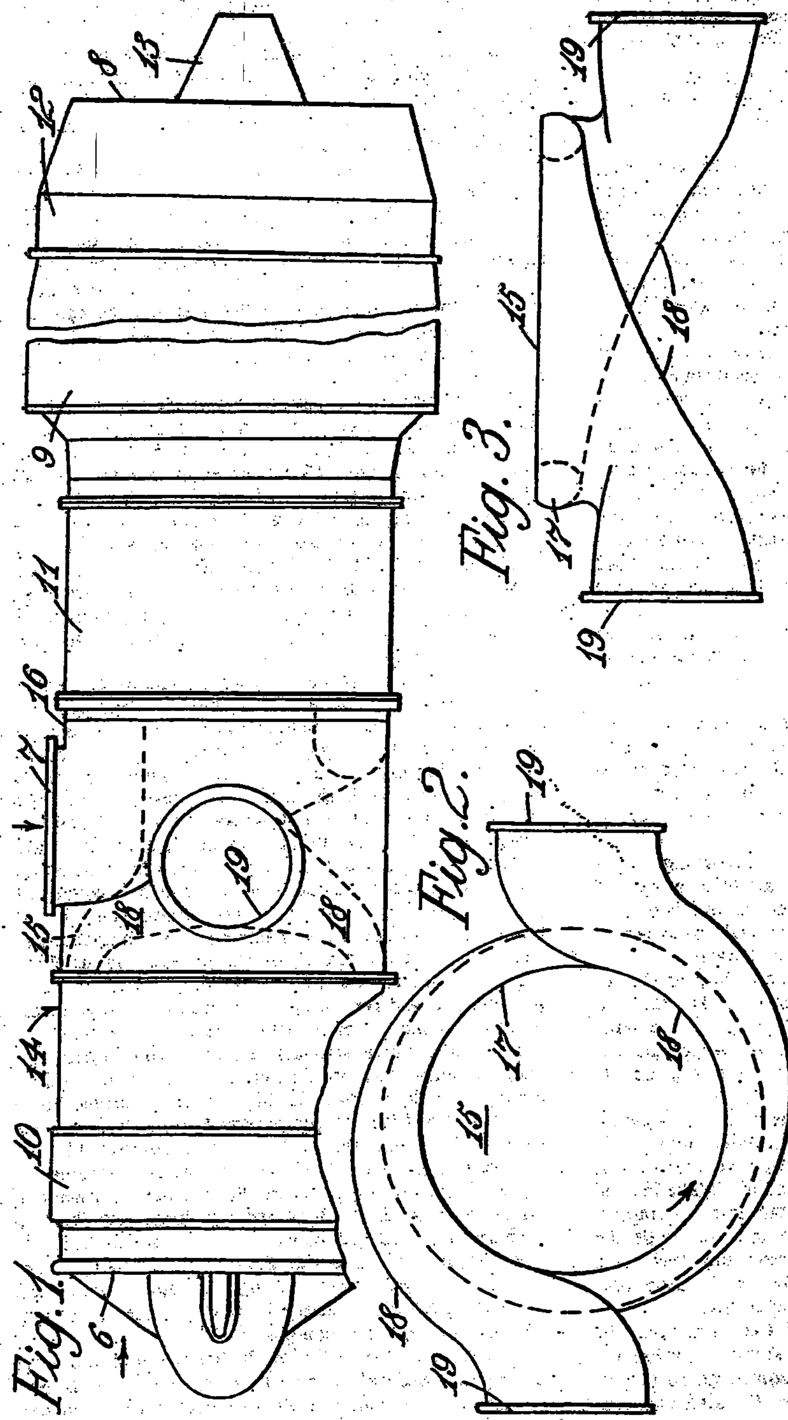

Fig. 1.

Fig. 2.

Fig. 3.

United States Patent Office 2,994,471
Patented Aug. 1, 1961

1

2,994,471

AIR SUPPLY UNIT

Gordon Manns Lewis and Peter Frederick Orchard, Bristol, England, assignors, by mesne assignments, to Bristol Siddeley Engines Limited, Bristol, England, a British company

Filed Mar. 10, 1958, Ser. No. 720,220
Claims priority, application Great Britain Mar. 19, 1957
4 Claims. (Cl. 230—116)

This invention relates to air supply units for supplying substantial quantities of compressed air, and concerns such units which are intended for use in aircraft, for example, for blowing air over the wing control surfaces of the aircraft. However, a unit according to the invention has other applications in and other uses than in aircraft, and the term "air" as used in this specification is intended to include any gaseous medium.

According to the invention an air supply unit comprises an axial flow load compressor, a gas turbine engine having an axial flow compressor, said engine being connected to drive said load compressor, and said load compressor being arranged coaxially with the axial flow compressor of said engine and on the side thereof remote from the turbine system of the engine, and deflection ducting connected to receive the discharge from said load compressor, and to deflect the discharge outwardly in a lateral direction with respect to the axis of the load compressor, said ducting being located between said load compressor and said engine.

According to a feature of the invention, the turbine system of the engine may comprise a single turbine, in which case the turbine is connected to drive the engine compressor and the load compressor.

When this feature is adopted it is preferred that the engine compressor and the load compressor have matching flow characteristics and are each connected to be driven directly by said turbine, for example through a common driving shaft. In this way the complication of reduction gearing is avoided.

The turbine system may, however, according to an alternative feature of the invention, comprise a pair of mechanically independent turbines one of which is connected to drive the engine compressor and the other of which is connected to drive the load compressor.

In the case of an air supply unit according to the invention intended for use in an aircraft, it is preferred that said engine is a gas turbine jet propulsion engine. The air supply unit may then be used to propel the aircraft as well as supply air for purposes which may include propulsion.

According to another feature of the present invention, the load compressor may have a straight annular air intake duct arranged co-axially with the load compressor, and the engine compressor may have an air intake located between said deflection ducting and the engine compressor. Where the unit is incorporated in an aircraft the air intake duct for the load compressor may lead from a forwardly facing annular air intake of the kind normally associated with the engine compressor.

According to another feature of the invention, the deflection ducting may comprise an annular portion co-axial with the load compressor and connected to receive directly through an annular outlet from the load compressor air compressed in the load compressor, at least one laterally directed discharge passage, and for each discharge passage a part spiral passage winding from said annular portion to the discharge passage in the direction of rotation of the load compressor. This arrangement of the ducting utilises the swirl of the air discharged by the load compressor to minimise the change of direction which the deflection ducting has to impart to the air

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discharging from the load compressor in order to deflect it outwardly in the lateral direction.

One embodiment of the present invention will now be described, merely by way of example, with reference to

FIGURE 1 is a diagrammatic side elevation of an air supply unit according to the invention intended for use in an aircraft,

FIGURE 2 is an end elevation on a larger scale of the deflection ducting, and

FIGURE 3 is a sectional elevation corresponding to FIGURE 2.

Referring to FIGURE 1, the unit comprises a gas turbine jet propulsion engine comprising an axial flow compressor 11, a combustion system 9 and a turbine system 12. The jet stream issuing from the turbine system emerges through an annular outlet 8 around an exhaust cone 13, and, when the unit is installed in an aircraft would be directed into a jet pipe terminating in a jet propulsion nozzle.

The air supply unit further comprises an axial flow load compressor 14 co-axial with the compressor 11 and on the side thereof remote from the turbine system 12 and deflection ducting 15 connected to receive the discharge from the compressor 14 and to deflect the discharge outwardly in a lateral direction with respect to the axis of the compressor 14. For convenience the term "laterally directed" will hereinafter be used to mean directed laterally with respect to the axis of the compressor 14, and in a similar way the term "axially directed" will be used to mean directed in the direction of the axis of the compressor 14.

The compressor 14 has a straight annular air intake duct 10 arranged co-axially therewith and leading from an axially directed air intake opening 6, and the the compressor 11 has a laterally directed air intake opening 7 opening into an axially directed annular air intake duct 16 co-axial with the engine compressor. The duct 16 is located between the deflection ducting 15 and the compressor 11. The turbine system 12 comprises only a single turbine, and the compressor 14 is coupled to rotate with the compressor 11 which is in turn connected to be driven by the turbine. The compressor 14 and the compressor 11 have matching flow characteristics and are each driven directly by the turbine. By matching the flow characteristics of the two compressors, the compressors can be driven at the same speed and the problems involved by the use of reduction gearing are avoided.

In an alternative arrangement, the turbine system 12 may comprise two mechanically independent turbines arranged in flow series, the high pressure turbine being connected to drive the compressor 11 and the low pressure turbine connected to drive the compressor 14. As will readily be understood the low pressure turbine receives as its working medium the combustion gases discharging from the high pressure turbine, and the two turbines are arranged co-axially with one another, the low pressure turbine driving the compressor 14 by means of a shaft which passes through a hollow drive shaft connecting the high pressure turbine with the compressor 11, and through the compressor 11.

The form of the deflection ducting 15 is shown more clearly in FIGURES 2 and 3 to which reference will now be made. The deflection ducting comprises an annular portion 17 co-axial with the compressor 14 and connected to receive directly through an annular outlet from the compressor 14 air compressed in the compressor 14. Winding from the annular portion 17 in the direction of rotation of the compressor 14 to two laterally directed discharge passages 19 located diametrically opposite one another are two part-spiral diffusers 18.

The direction of winding of the part-spiral diffuser

RMD AS 000115

portions 18 is made the same as the direction of rotation of the compressor 14 so as to match the direction of swirl of compressed air discharged from the compressor 14 in order to minimise the deflection which has to be imparted to the discharge by the walls of the diffuser portions in order to deflect the discharge outwardly through the laterally directed discharge passages 19.

In an alternative arrangement, diffuser portions 18 may be dispensed with, and an annular diffuser may be inserted co-axial with the compressor 14, the annular diffuser being connected to receive the discharge of the compressor 14 directly through said annular outlet. In this case the deflection ducting would comprise an annular portion such as 17 communicating with the downstream end of the annular diffuser and one or more laterally directed discharge passages such as 19 communicating directly with the annular portion. The advantage of the part-spiral diffuser arrangement illustrated however is that it is more compact, and does not occupy so much axial space as the alternative arrangement.

The annular air intake duct 16 and the annular portion 17 of the deflection ducting 15 surround the drive shaft of the compressor 14.

Instead of providing two discharge passages 19 and two part-spiral portions 18, there may be a single discharge passage 19 connected with the portion 17 by a single part-spiral passage 18, and furthermore, the single passage 19 may be directed oppositely to the air intake opening 7 instead of at right angles thereto.

An axial flow gas turbine jet propulsion engine may readily be converted into an air supply unit as described, by inserting between the air intake casing of the engine and the compressor casing of the engine a load compressor section, and a section comprising deflection ducting such as 15 and an alternative air intake arrangement for the compressor of the engine, the engine air intake being used to feed the load compressor instead of the engine compressor, and the load compressor being connected to be driven with the engine compressor by an extension drive shaft on the engine compressor.

We claim:

1. An air supply unit comprising a gas turbine engine including an axial flow engine compressor having an air inlet open directly to atmosphere, combustion equipment connected to receive compressed air from the compressor, and a turbine system connected to receive the products of combustion from the combustion equipment, an axial flow load compressor arranged coaxially with the engine compressor and on the side thereof remote from the turbine system of the engine, coupling means drivingly connecting the engine to said load compressor, said load compressor having an annular outlet duct between the

engine compressor and the load compressor and encircling the engine axis, and deflection ducting providing a discharge outlet for air compressed in the load compressor, said deflection ducting forming a part-spiral passage which is connected to receive compressed air from the outlet duct of the load compressor and which winds from said annular outlet duct to said discharge outlet in the direction of rotation of the load compressor.

2. An air supply unit as claimed in claim 1, wherein the load compressor has a co-axial straight annular air intake duct leading from an axially directed air intake opening for the load compressor, and the engine compressor has a laterally directed air intake aperture open directly to atmosphere and leading into an axially directed annular air intake duct for the engine compressor located between said deflection ducting and the engine compressor and co-axial with the engine compressor.

3. An air supply unit as claimed in claim 1, wherein said part-spiral passage is shaped to act as a diffuser.

4. An air supply unit comprising a gas turbine engine including an axial flow engine compressor having an air inlet open directly to atmosphere, combustion equipment connected to receive compressed air from the compressor, and a turbine system connected to receive the products of combustion from the combustion equipment, an axial flow load compressor arranged coaxially with the engine compressor and on the side thereof remote from the turbine system of the engine, said load compressor having an annular outlet duct which is disposed between the engine compressor and the load compressor and which encircles the engine axis, deflection ducting providing a discharge outlet for air compressed in the load compressor, said deflection ducting forming a part-spiral passage which is connected to receive compressed air from the outlet duct of the load compressor and which winds from said annular outlet duct to said discharge outlet in the direction of rotation of the load compressor, and a drive shaft drivingly connecting the engine to said load compressor, said drive shaft being encircled by the deflection ducting.

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PATENT SPECIFICATION

1,021,797

DRAWINGS ATTACHED.

1,021,797

Date of Application and filing Complete Specification: *March 4, 1964.* *No. 9221/64.*

Application made in Netherlands (No. 289,828) on March 6, 1963.

Complete Specification Published: March 9, 1966.

Index at Acceptance:—G3 P(1B, 1C, 4, 16B2, 24K3, 24KX); F1 C(2J1A, 2J1D, 2J2J1).

Int. Cl.:—G 05 b, c, d /F 04 d.

COMPLETE SPECIFICATION.

Method and Apparatus for the Protection of a Centrifugal Compressor.

We, SHELL INTERNATIONALE RESEARCH MAATSCHAPPIJ N.V., a Company organised under the Laws of the Netherlands, of 30 Carel van Bylandtlaan, The Hague, The Netherlands, do hereby declare the invention, for which we pray that a patent may be granted to us, and the method by which it is to be performed, to be particularly described in and by the following statement:—

The invention relates to a method and apparatus for the protection of a centrifugal compressor in order to prevent entirely or almost entirely the so-called surging of the compressor. Surging may occur in a compressor when, at a certain pump pressure, the gas velocity through the compressor becomes too low, or when, at a certain gas velocity the pressure becomes too high. In order to counteract this surging the compressor is provided with a by-pass or a blow-off line, which during normal operation (i.e. when the gas load is sufficiently high), is closed by a control valve. As soon as the risk or surging arises, the valve is opened to a greater or lesser extent, so that the gas flow through the compressor increases and surging is avoided. Since the by-passing or the blow-off of gas through the by-pass or the blow-off line represents a loss, in any case a loss of power, the valve is only opened when and insofar as it is necessary to avoid surging.

It has already been proposed to operate the control valve by a control system, to which signals dependent on the pressure level of the compressor and of the gas flow through the compressor are supplied. Thus, for example according to U.S. patent specification 2,000,721, a signal proportional to the discharge pressure of the compressor, is compared in the controller, which controls the position of the control valve, with the sum of two signals, one of which depends on the flow rate of the gas feed and the other being dependent on the flow rate of the gas stream in the by-pass.

This control system can at best only work well at a single pressure level, i.e. the pressure level at which the controller has been set. If the suction pressure varies, it would be necessary, in order to obtain an effective protection against surging, continually to alter the setting of the controller, this being almost impracticable.

Hence, when the inlet pressure of the gas to be compressed is not constant or substantially constant, which often occurs when a plurality of compressors are connected in series, no effective protection of the compressor against surging is possible.

The invention now provides a method by means of which an effective protection against surging is obtained, which is practically independent of the pressure level at which the compressor operates. Moreover the protection is likewise independent or substantially independent of the inlet temperature of the gas to be compressed.

Special embodiments of the protection according to the invention also afford the possibility of making the compressor operate safely without or substantially without loss of power almost up to the range in which surging may occur. This is particularly important when the gas load of the compressor varies considerably. It is then particularly advantageous, when the range within which the gas load can vary without loss is as wide as possible. However, when, as the result of a gas load which is too low (or pressure which is too high), operations cannot be conducted without loss, the loss is kept at a minimum while at the same time a stable control of the control valve is ensured.

According to the present invention the throughput of the control valve is controlled by the output signal of a controller which

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compares a set value with input signal being the quotient of two signals one of which (Δp) is dependent on the gas flow through the compressor and the other (p) is dependent on the pressure level of the compressor, the direction of action of the control being such that—when $\frac{\Delta p}{p}$ or $\frac{p}{\Delta p}$ attains a value which is respectively smaller or greater than the set value of the controller—the controller produces an output signal in the valve-opening sense.

The signal dependent on the gas flow through the compressor is as a rule a differential pressure signal measured across an orifice, a venturi or a similar device.

Preferably the signal dependent on the gas flow through the compressor is measured at the discharge side of the compressor. The signal dependent on the pressure level is preferably obtained by measuring the discharge pressure of the compressor. Good results are obtained by a combination of Δp measured at the suction side of the compressor (Δp_1) with p measured at the discharge side of the compressor (p_2), or vice versa (Δp_2 and p_1). In practice the best control is achieved by measuring both Δp and p at the discharge side of the compressor (thus with the combination of Δp_2 and p_2).

By changing the set value of the controller, the moment at which the control valve begins to open may be brought closer to the surge limit of the compressor or may be further removed therefrom.

The method according to the invention is particularly useful when the gas load is normally well outside the danger area (this is the area in which surging of the compressor would occur if the control valve were closed) and does not vary very much. In this case the controller is usually provided with proportional action only and moreover with a wide proportional band. The wide proportional band is necessary for a stable control. If an unexpected deviation from the normal state now occurs, for example because the discharge of compressed gas becomes blocked, the controller prevents surging.

However, when the normal operation of the compressor entails a considerable variation in the load and thereby comes close to or exceeds the surge limit (so that the control valve has to act repeatedly or is permanently in action in order to avoid surging) the protection by means of a controller with proportional action only is less satisfactory: as a result of the relatively great width of the proportional band of the controller, the control is not very economic. If, however, an improvement in the economy were attempted by the introduction of integral action, the compressor may run risks when there are rapid variations in the load.

According to another characteristic of the invention it is therefore desirable to use two controllers connected in parallel, one of which has proportional action only and moreover a relatively narrow proportional band, while the other has both proportional and integral action and moreover a relatively wide proportional band; the output signals of these controllers are passed to an auxiliary relay, which causes the control valve to be controlled only by that signal which at any moment would impart the greatest throughput to the control valve.

As a rule both controllers will be connected to the same dividing circuit or relay (circuit or relay which supplies the quotient $\Delta p/p$ or $p/\Delta p$); in principle, however, it is possible to use various dividing circuits or relays, in order to determine and use various quotients (for example $\frac{\Delta p_2}{p_2}$ and $\frac{\Delta p_2}{p_1}$).

A similar result may be achieved by using a single controller, which in addition to proportional and integral action also has derivative action and preferably a wide proportional band.

The protection system according to the invention may be carried out in various ways, for example hydraulically, pneumatically or electrically; a pneumatic embodiment is often used. If a pneumatic embodiment is employed the output signal of the auxiliary relay (which is used in the control system having two controllers)—if necessary after amplification—may be used in order to feed that part of the controller, which produces the integral action, with which one of the controllers is provided.

Further, an apparatus may be used in a manner otherwise known per se, which prevents the controller from becoming saturated as a result of the integral action (so-called "anti reset wind-up").

The invention may be carried into practice in various ways but it will now be further illustrated by way of example with reference to the accompanying drawings in which:

Figure 1 relates to the use of a single controller and

Figure 2 to the use of two controllers connected in parallel.

Figure 1 shows diagrammatically a method for the compression of natural gas by means of a centrifugal compressor 1. The gas is supplied through a line 2 and leaves the compressor through a line 3. The compressed gas is subsequently cooled by means of a cooler 4 and is freed in a separator 5 from any condensate which may have been formed. The condensate is discharged through a line 6 and the compressed gas through lines 7 and 8, after which it may be pumped back to the earth formation or passed to an installation where liquid methane is prepared. A gas

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by-pass 9 connects the lines 7 and 2; a control valve 10, which is normally closed, is incorporated in this by-pass. If surging of the compressor occurs or if there is a danger of surging, the control valve is opened so that surging is eliminated or prevented. The by-pass 9 could have been connected to the discharge side of the compressor at an earlier point, for example, just before the cooler 4.

In the line 3 an orifice 11 is inserted (which may be a venturi or similar device) which is connected to a differential pressure meter 12. This meter produces a signal (Δp_2), which is proportional to the differential pressure occurring across the orifice. A pressure meter 13 is likewise connected to the line 3 and produces the signal (p_2) which is proportional to the discharge pressure of the compressor.

The signals of the meters 12 and 13 are passed to a dividing circuit or relay 14, the output signal of which is proportional to $\frac{\Delta p_2}{p_2}$. This signal is supplied to a controller 15 which at 16 may be set at a certain value of a quotient (A) which is otherwise adjustable according to magnitude. The controller is preferably only provided with proportional action. The controller 15 compares the measured value (originating from the dividing relay 14) of the quotient $\frac{\Delta p_2}{p_2}$ with the set value A. The output signal 17 of the controller 15 controls the control valve 10. If $\frac{\Delta p_2}{p_2} \geqslant A$ the control valve 10 is or remains closed. When, however, $\frac{\Delta p_2}{p_2} < A$ (for example, owing to the fact that no gas or less gas than usual is withdrawn at 8) the control valve is opened to a greater or lesser degree, in any case to such an extent that no surging of the compressor occurs. By altering the set value A, the moment at which the control valve 10 begins to open is shifted.

This protection of the compressor has the advantage of being practically independent of the compressor pressure. When there is a change in the pressure level, for example when there is a rise or fall in the pressure of the natural gas fed, the protection against surging continues to operate without anything having to be changed.

In the pneumatic embodiment the signal pressure (17) is in general so chosen that when the signal pressure disappears the valve 10 is opened.

The quotient $\frac{p_2}{\Delta p_2}$ could have been used instead of the quotient $\frac{\Delta p_2}{p_2}$ and the former could have been passed to the controller 15. The set value then changes, as does the direction of the control. In this case the valve is, of course, opened as soon as $\frac{p_2}{\Delta p_2}$ becomes greater than the set value A^1 where $A^1 = \frac{1}{A}$.

Figure 2 shows the changes which are necessary when using two controllers 15*a* and 15*b* connected in parallel and both connected to the same dividing relay 14. The controller 15*a* only has proportional action and is moreover adjusted to at a relatively narrow proportional band (for example 10%). The controller 15*b* has both proportional and integral action and moreover a wide (compared with controller 15*a*) proportional band (for example 100 to 250%). The set values 16*a* and 16*b* of the controllers respectively, differ slightly, in such a way that the controller 15*b* can first come into action. The control range of the controller 15*a* lies on that side of the operating line of the controller 15*b* where the surge limit also lies and usually near to the said operating line (at the operating line applies: $\frac{\Delta p}{p} = A$).

The output signals of the controllers are now passed to an auxiliary relay 18 which only allows that signal to pass through which at any moment would impart the greatest throughput to the control valve 10. In the present embodiment (in which the control valve closes when the pneumatic signal 17 increases in magnitude) this means that the relay 18 continually transmits the smaller of the two signals from the controllers 15*a*, 15*b*.

The part 16*c* of the controller 15*b* which relates to the integral action thereof is preferably fed by the output signal of the relay 18. In Figure 2 the signal 17 is therefore passed to the said part via an amplifier 19. This results in a more rapid operation of the integral action. Further an apparatus is generally used which prevents the controller from becoming saturated as a result of the integral action (anti reset wind-up).

If the load of the compressor 1 (in so far as it concerns the discharge at 8) approaches the surge limit, the controller 15*b* first comes into action since the relevant operating line is the first to be overshot. Since the operating line of controller 15*b* can be placed relatively close to the surge limit the operation of the compressor installation remains economic; for losses as a result of by-passing of gas through the by-pass do not occur as long as the operating line of controller 15*b* (in so far

as it concerns the discharge at 8) is not overshot, since the control valve is then closed; and the control valve is only slightly opened when the operating line is only slightly overshot, so that the losses are small.

If, however, a further change in the load brings the compressor closer to or in the danger area, the other controller 15a can immediately come into operation and open the control valve forthwith. Without this controller 15a there is, however, a great risk of surging of the compressor, since the controller 15b, as a result of the wide proportional band and the action resulting from the integral action, cannot open the control valve 10 in time.

On the other hand a permanent unstable control (owing to the narrow proportional band of controller 15a) need not be feared, since eventually the controller 15b reassumes control, viz. as soon as the integral action has been able to build up a signal of suitable magnitude.

Finally, the integral action also ensures that, when the compressor must operate permanently with a more or less widely opened control valve, no difference between measured value and set value occurs which could otherwise lead to surging of the compressor.

WHAT WE CLAIM IS:—

1. A method of preventing surging of a centrifugal compressor wherein a by-pass or blow-off line for the gas is provided with a control valve, the throughput of which is controlled by the output signal of a controller which compares a set value with an input signal being the quotient of two signals one of which (Δp) is dependent on the gas flow through the compressor and the other (p) is dependent on the pressure level of the compressor, the direction of action of the control being such that—when $\frac{\Delta p}{p}$ or $\frac{p}{\Delta p}$ attains a value which is respectively smaller or greater than the set value of the controller—the controller produces an output signal in the valve-opening sense.

2. A method as claimed in Claim 1 wherein the signal dependent on the gas flow through the compressor is measured as a differential pressure signal across an orifice, a venturi or a similar device.

3. A method as claimed in Claims 1 or 2, wherein the signal dependent on the gas flow through the compressor is measured at the discharge side of the compressor.

4. A method as claimed in any one of the preceding claims wherein the signal dependent on the pressure level is obtained by measuring the discharge pressure of the compressor.

5. A method as claimed in any one of the preceding claims wherein the controller has proportional, integral and derivative action.

6. A method as claimed in any one of Claims 1 to 4, wherein two controllers connected in parallel are used, one of which has only proportional action and moreover a relatively narrow proportional band, while the other has both proportional action and integral action and in addition a relatively wide proportional band; and that the output signals of these controllers are passed to an auxiliary relay, which causes the control valve to be controlled only by that signal which at any moment would impart the greatest throughput to the control valve.

7. An apparatus suitable for the protection of a centrifugal compressor in the manner described in any one of the preceding claims, comprising a pressure meter for the pressure level of the compressor and a differential pressure meter for the gas flow through the compressor, a dividing relay which is connected to receive the outputs of these meters, a controller which is connected to receive from the dividing relay an output comprising the value of the quotient of differential pressure and pressure and to compare this quotient with the set value and a connection for connecting the output of the dividing relay to a control system of the control valve.

8. An apparatus as claimed in Claim 7, in which the controller has proportional, integral and derivative action.

9. An apparatus as claimed in Claim 7, including two controllers both of which are connected to a dividing relay, one of these controllers having only proportional action and moreover a relatively narrow proportional band, the other having both proportional and integral action and in addition a relatively wide proportional band; a connection between the output of each of the controllers and an auxiliary relay which is capable of selecting from the two signals supplied thereto that signal which would impart at any moment the greatest throughput to the control valve, and by a connection for connecting the output of the auxiliary relay to the control system of the control valve.

10. An apparatus as claimed in Claim 9 including a connection from the integrating part of the controller having integral action, if necessary via an amplifier, to the output of the auxiliary relay.

11. An apparatus as claimed in Claim 9 or 10, characterized by means for avoiding saturation of the controller by the influence of the integral action.

12. A method of preventing surging of a centrifugal compressor substantially as described herein with reference to Figure 1 or Figure 2 of the accompanying drawings.

13. Apparatus for preventing surging of a

centrifugal compressor substantially as described herein with reference to Figure 1 or Figure 2 of the accompanying drawings.

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Abingdon: Printed for Her Majesty's Stationery Office, by Burgess & Son (Abingdon), Ltd.—1966.
Published at The Patent Office, 25 Southampton Buildings, London, W.C.2,
from which copies may be obtained.

1021797 COMPLETE SPECIFICATION

1 SHEET *This drawing is a reproduction of the Original on a reduced scale*

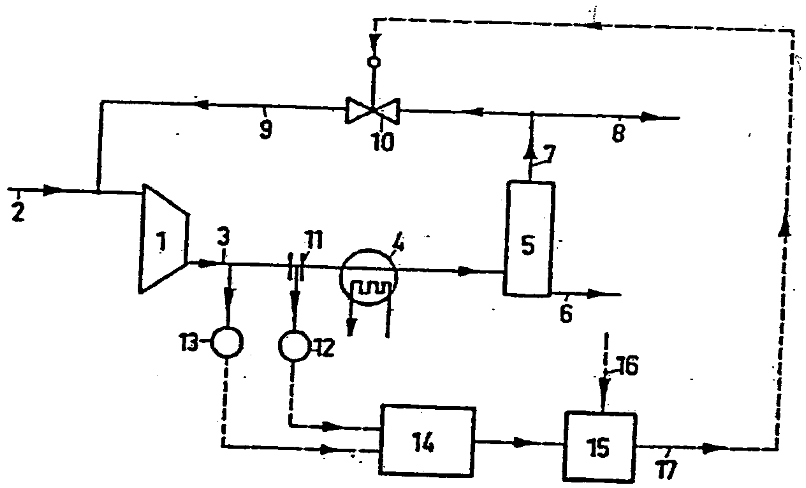

FIG.1

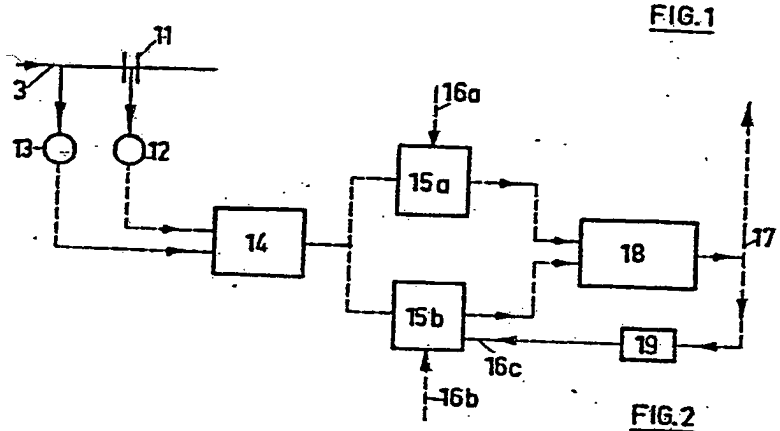

FIG.2